United States Patent

Cline

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[54] **APPARATUS FOR CONTROLLING THE CHARACTERISTICS OF FLUID PRESSURE OPERATED FRICTION TYPE POWER ABSORPTION DEVICES**

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Related U.S. Application Data

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[52] **U.S. Cl.** **73/135**, 73/117

[51] **Int. Cl.** **G01l 3/16**

[58] **Field of Search** 73/117, 134, 135; 318/304, 318/372; 188/180, 182

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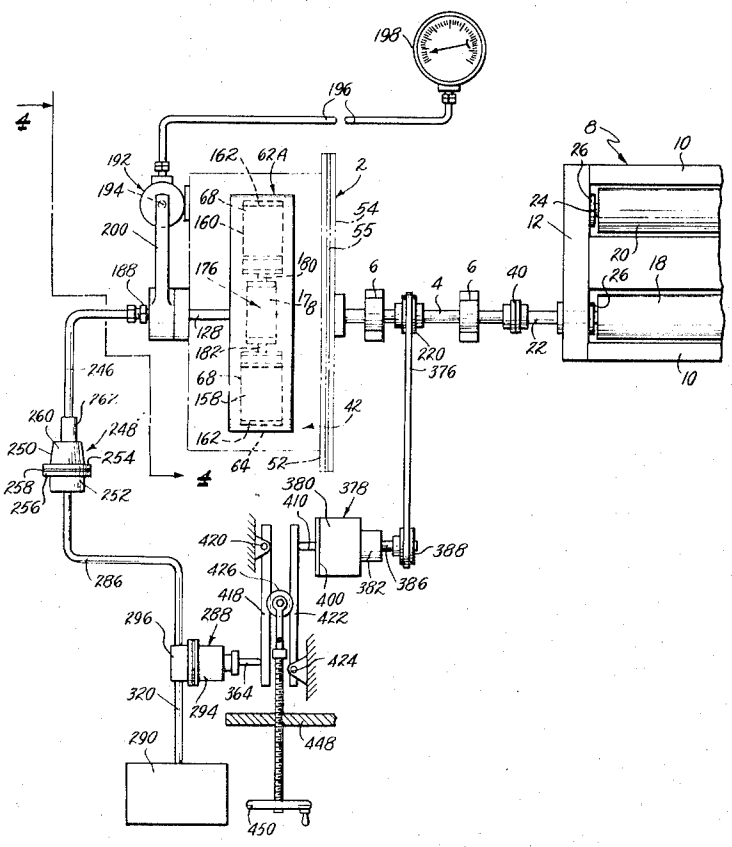
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[57] **ABSTRACT**

Apparatus for controlling a rotary power absorber while absorbing driving torque produced by a prime mover. The power absorber includes a housing containing fluid pressure operated brake elements that are actuatable hydraulically to provide retarding force. The control means for the brake applying means includes a governor-controlled air valve and a transducer which converts air pressure into hydraulic pressure for actuating the brake elements. The governor is driven at a speed proportional to the speed of rotation of the prime mover and actuates the air valve so that fluid pressure is applied to the brake elements in direct proportion to the speed of the prime mover. An adjustable linkage is interposed between the governor and the air valve for controlling the maximum retarding force at various engine speeds. The linkage includes two parallel bars pivoted at opposite ends, with the governor acting on the free end of one bar and the free end of the other bar acting on the air valve. A roller is positionable at any desired point along the length of the bars to vary the ratio of air valve movement to governor movement, as desired, to correspondingly vary the retarding force. The retarding force is increased and decreased at a rate faster than the changes in speed and torque of the prime mover to avoid stalling of the prime mover as frequently occurs when a constant load is sought to be applied to the prime mover and there is a momentary failure in power of the prime mover.

12 Claims, 7 Drawing Figures



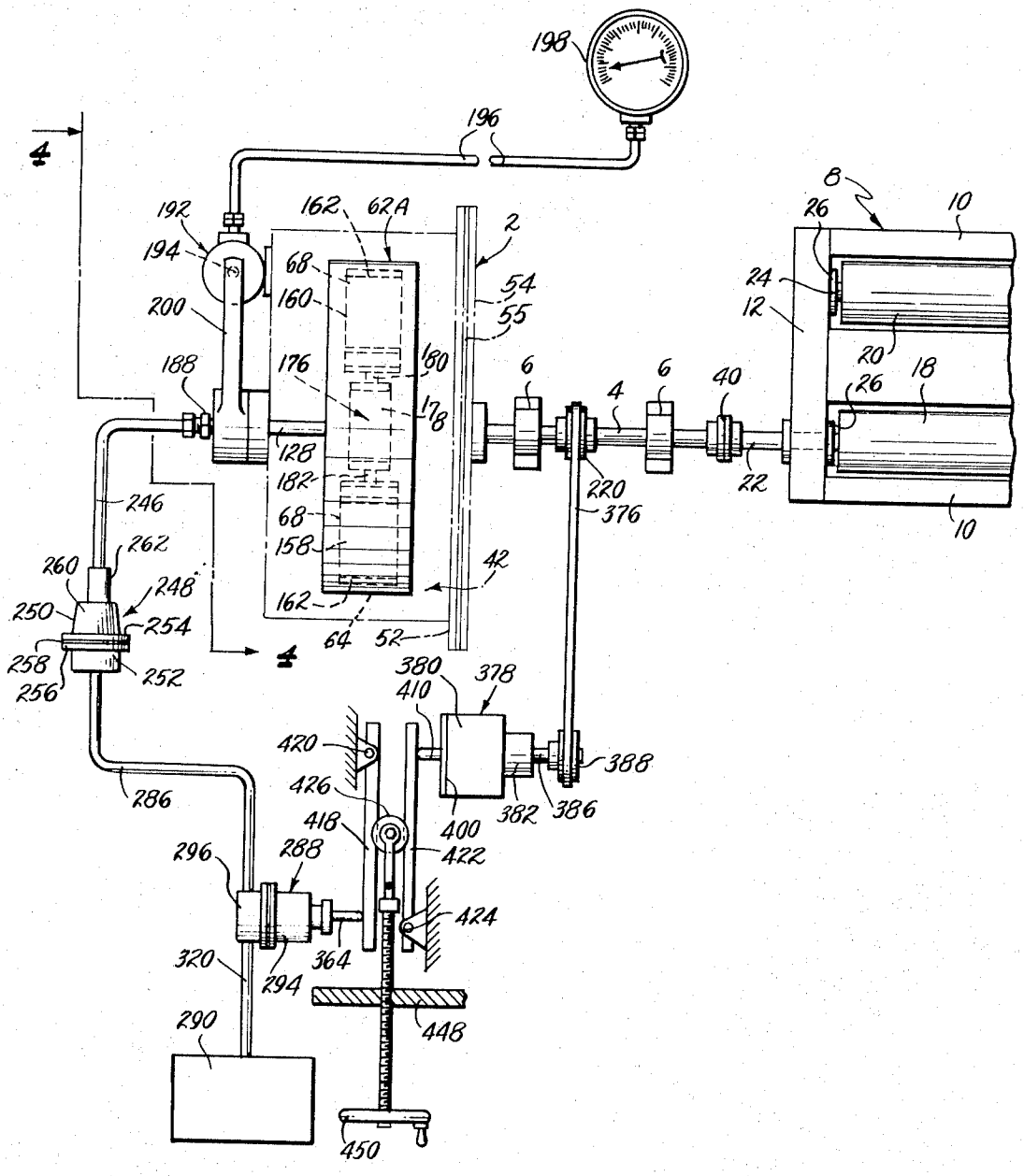


Fig. 1.

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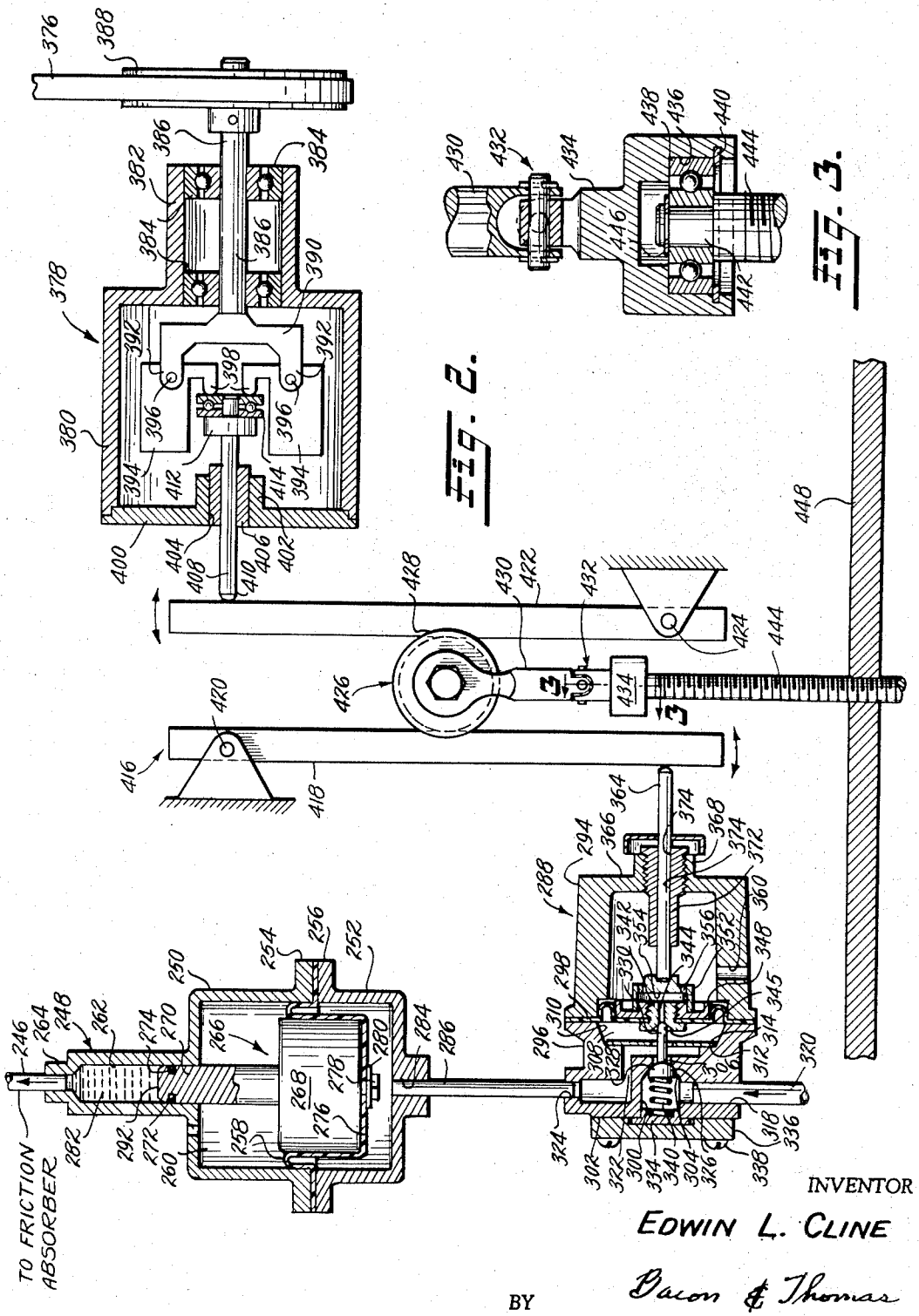
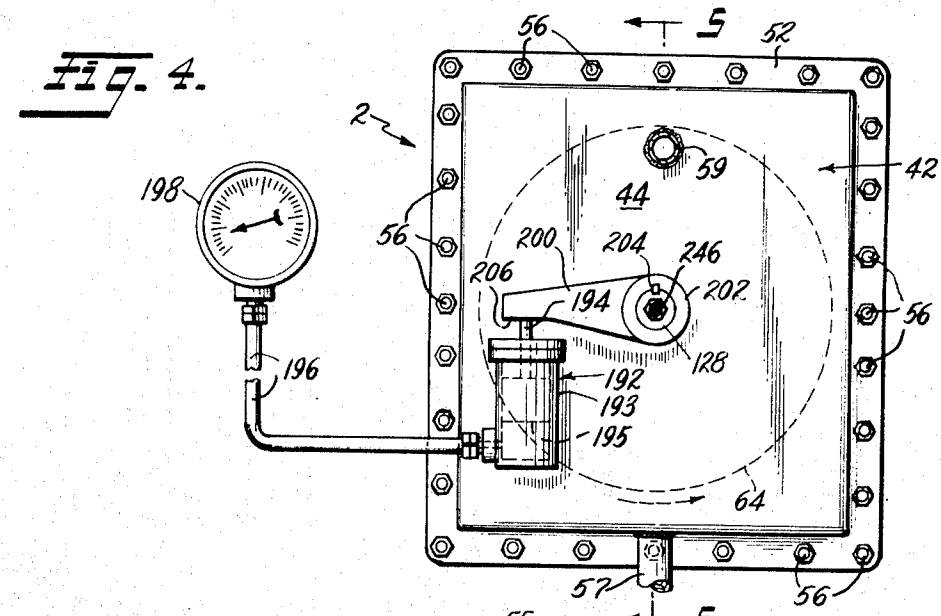

TO FRICTION ABSORBER
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FIG. 2.

FIG. 3.

Fig. 4.



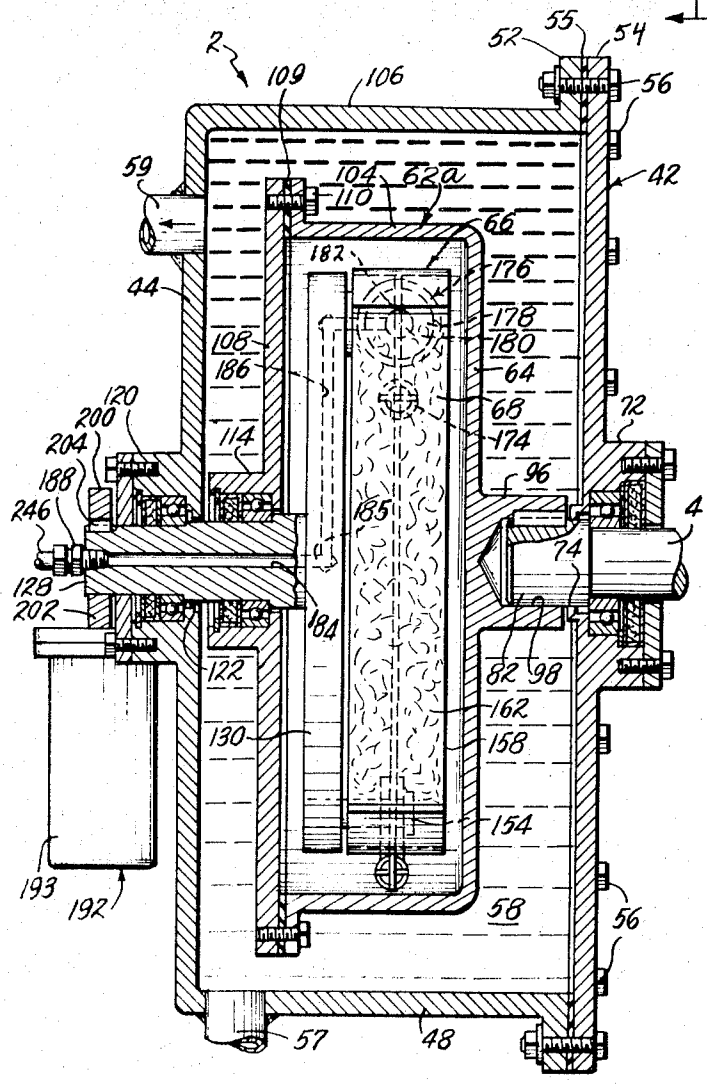


Fig. 5.

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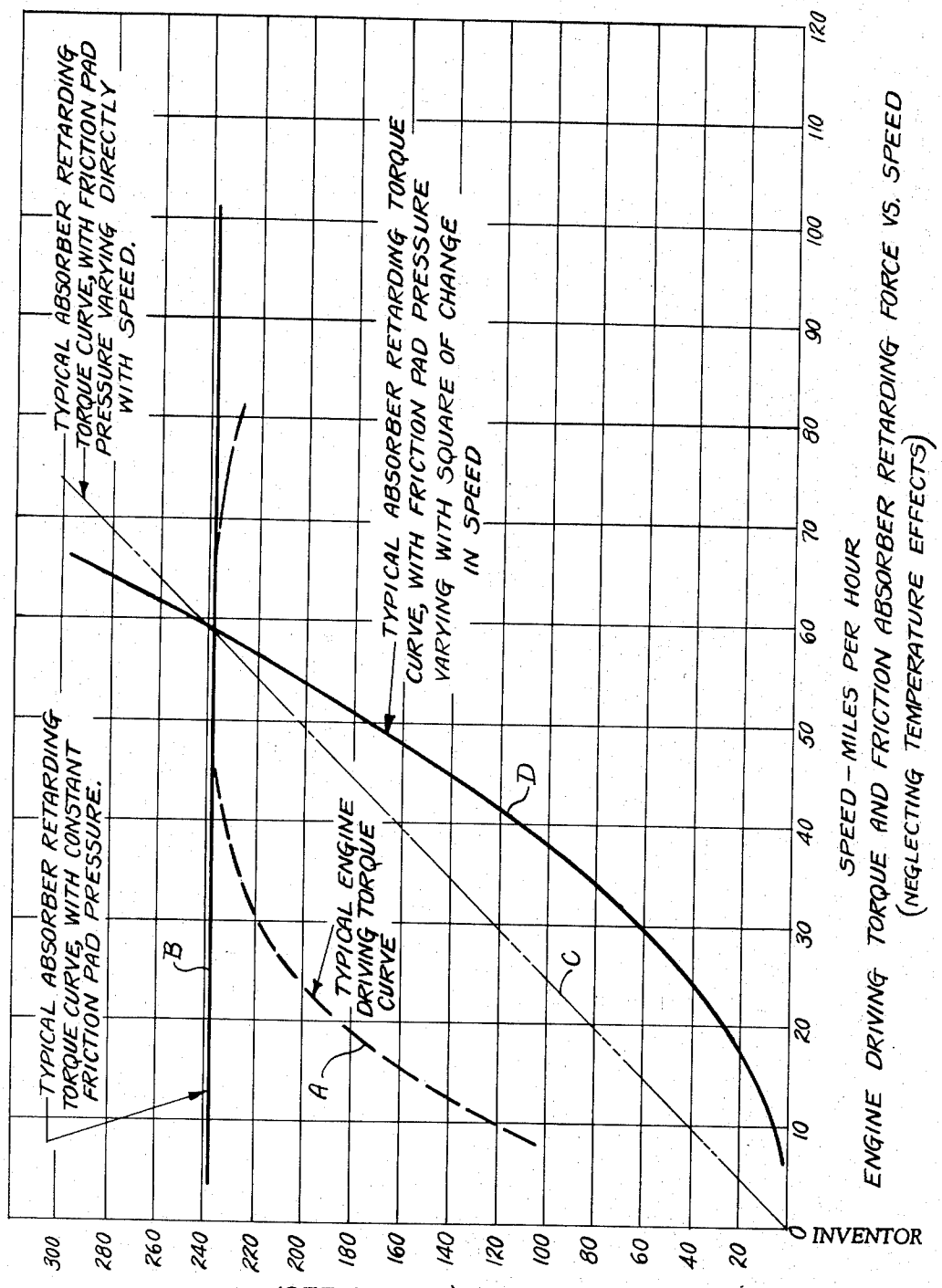


FIG. 6.

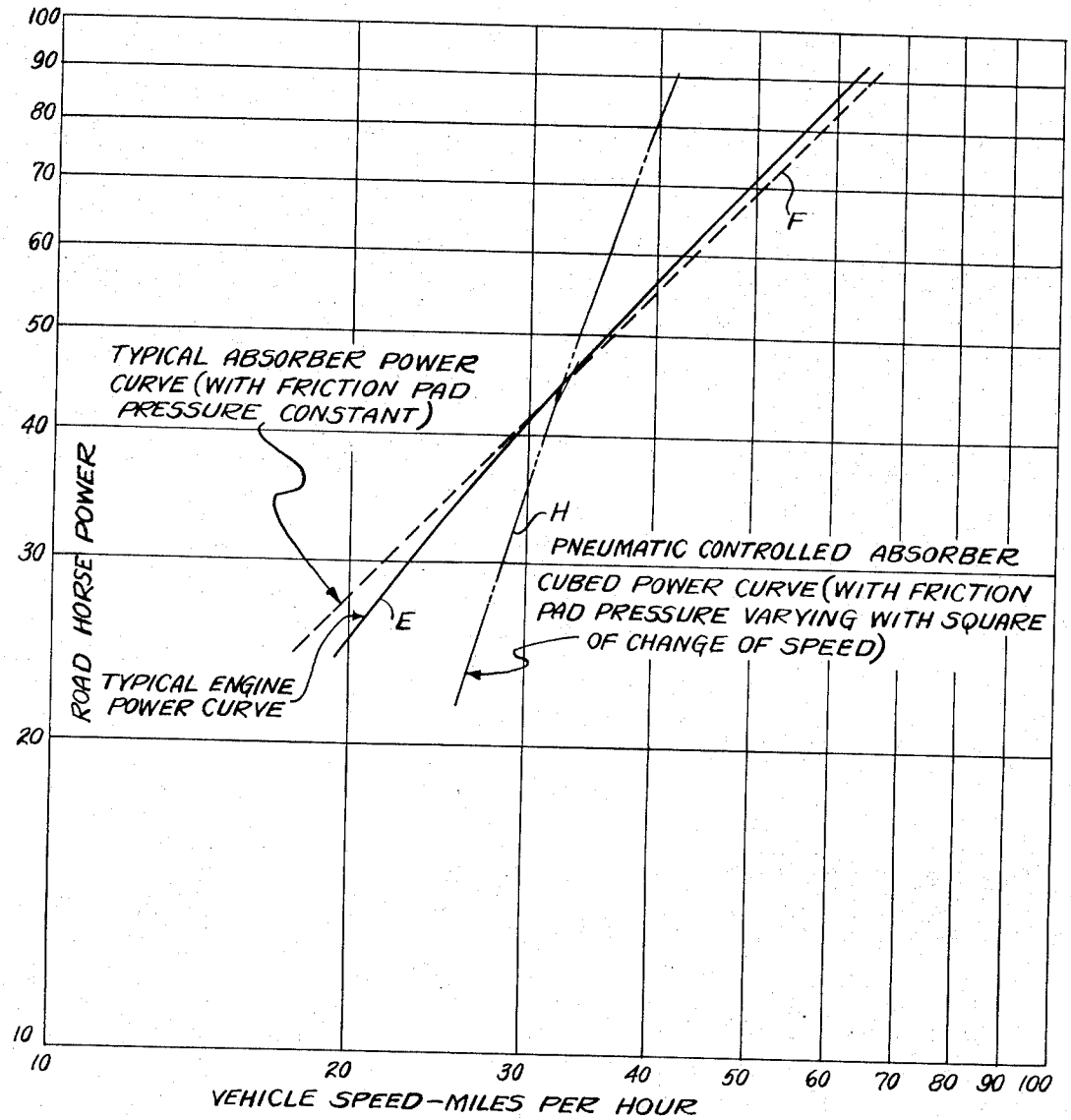
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ENGINE AND FRICTION ABSORBER POWER VS. SPEED.
(NEGLECTING TEMPERATURE EFFECTS)

Fig. 7.

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APPARATUS FOR CONTROLLING THE CHARACTERISTICS OF FLUID PRESSURE OPERATED FRICTION TYPE POWER ABSORPTION DEVICES

CROSS-REFERENCE

This application is a division of my co-pending application Ser. No. 559,490 filed July 22, 1966 now U.S. Pat. No. 3,453,874.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to equipment for testing the under-load performance characteristics of a prime mover by rotary friction type power absorption devices, and more particularly to novel load control means therefor. For convenience, such power absorption device will hereinafter simply be referred to as a friction absorber.

2. Description of the Prior Art

Currently available friction absorbers comprise a rotating brake drum or disk to be connected with the output shaft of the prime mover, and stationary friction pads or brake shoes that are engageable with the drum or disk to apply a retarding force thereto by frictional contact. The degree of retarding action is dictated by the force with which this frictional contact is made. The system that is used to apply this force is referred to as the "Load Control System." In simple friction absorbers this is a "fixed" force and only one speed versus power curve is possible for the reason that a constant retarding force is applied regardless of the speed or torque that is developed by the engine. In more flexible friction absorbers, the load control will allow the operator to manually vary the force of frictional contact. In the case of a hydraulically actuated system, for example, this is accomplished by the operator varying the fluid pressure. A series of speed versus power curves can then be obtained.

Due to the speed versus power characteristics of friction absorbers, the "fixed" force load control system is unsatisfactory because, for a given contact pressure of the brake shoes, the retarding force remains constant (neglecting the effects of temperature) and the same amount of torque will be absorbed throughout the speed range. Friction absorbers that produce a constant retarding force are further objectionable, in that such devices will cause the prime mover to stall in the event that the engine should momentarily miss or lose power for any reason. Such constant force friction absorbers are still further objectionable from the standpoint that they are unstable over the range in which the retarding force is equal to engine torque, which may cover a substantial speed range.

Since horsepower involves both speed and torque, the horsepower will increase and decrease directly with speed. In testing engines with friction absorbers, speed stability can be acquired only when the retarding force of the friction absorber increases and decreases with speed faster than that of the prime mover being tested. An ideal condition would be for the friction retarding force to start at zero with zero speed and change as a square of the speed change. This relationship is very close to the load imposed on a conventional automobile engine when the vehicle is driven on a level road and, hence, represents highly desirable loading characteristics to be simulated in practice. Such operational characteristics obviously cannot be attained with the "fixed" force load control for reasons stated above. Likewise, it is extremely difficult and practically impossible to establish and maintain such operating characteristics in a friction absorber by manual control of the force load, and, hence, such manual control leaves much to be desired.

SUMMARY OF THE INVENTION

Accordingly, there has long existed the need for load control means for friction absorbers that will render the same practical and avoid the principal objections thereto noted above. The load control means of the present invention is designed to obviate these objections and may take any number of forms. Each load control means is made to simulate road conditions, to maintain stability at any speed, and to control the friction absorber means so that the retarding force of the friction absorber means is zero at zero engine speed and will rise and fall faster than the torque of the prime mover being tested. The present load control means contemplates a fluid control system that is responsive to engine speed, and which can be adjusted and the load pre-selected to impose a retarding force of a given value at a given speed and automatically and correspondingly control the operating characteristics of the friction absorber in accordance with the foregoing relationship at all other speeds. The system is also capable of being remotely controlled by an operator.

The present load control means is applicable in principle to all types of friction absorbers, irrespective of whether the friction absorber is directly or indirectly coupled with the output shaft of the prime mover. A direct method would be to connect the prime mover shaft directly to the input shaft of the friction absorber, as in an engine dynamometer test stand setup. An indirect method would involve the incorporation of the friction absorber in a chassis dynamometer for testing engines of automobiles or trucks without removing the engine from the vehicle. In such case, rolls are usually provided to form a type of treadmill for the drive wheels of the vehicle and the friction absorber is then connected with a driven roll. Power from the engine would then be normally transmitted to the friction absorber through the vehicle transmission and differential. For illustrative purposes, and not by way of limitation, the load control means of the present invention is shown and described in connection with a friction absorber associated with a chassis dynamometer.

More specifically, the invention comprises a friction power absorber including a housing containing fluid pressure operated friction brake elements that are actuatable to provide retarding force. The control means for the brake applying means includes a governor driven at a speed proportional to the speed of rotation of the prime mover so that linear movement of an element of the governor is produced in direct proportion to the speed of the prime mover. Such movement is transmitted through an adjustable linkage system to an air pressure control valve, which activates a transducer to apply hydraulic pressure to the brake elements for controlling the maximum brake retarding force at various engine speeds. The retarding force is increased and decreased at a rate faster than the changes in speed and

torque of the prime mover. The brake actuating pressure can be varied as the square of the speed of the prime mover, or any other suitable mathematical function of the speed of the prime mover.

Accordingly, the principal object of the present invention is to provide a load control apparatus for controlling the retarding force characteristics of a friction absorber, so that the retarding force produced thereby increases and decreases in value at a rate faster than the increases and decreases in driving torque applied to said absorber from a prime mover, and so that the applied retarding force value for any given prime mover speed can be pre-selected at will.

Another object is to provide a dynamometer, including a rotary friction absorber and control means for creating retarding force in opposition to driving torque applied thereto from a prime mover, designed so that the value of the retarding force will be changed in proportion to the occurring changes in driving torque and speed applied to said friction absorber by the prime mover.

A further object is to provide adjustable control means for a friction absorber, constructed so that various value relationships between retarding force and driving torque can be pre-selected.

Another object is to provide apparatus for controlling the friction characteristics of a friction absorber constructed to automatically vary the retarding force produced by the friction absorber in a preselected manner and as a function of the speed of rotation of the shaft supplying driving torque to said unit.

A more specific object is to provide load control means for a friction absorber, wherein means responsive to the speed of the prime mover being tested is utilized to regulate a fluid pressure control system to actuate a power absorber in accordance with speed changes of the prime mover, to correspondingly vary the retarding force created by said friction absorber to provide a pre-selected load on the prime mover at a given speed.

Another object is to provide load control means for a friction absorber that can be remotely controlled and which allows pre-selection of the degree of load to be applied to a prime mover.

Other objects and many of the attendant advantages of the invention will become readily apparent from the following description, when taken together with the accompanying drawings.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a fragmentary, diagrammatic plan view of a chassis dynamometer incorporating a friction absorber controlled by a pneumatic-hydraulic pressure transducer, an air control valve for supplying air under pressure to said transducer, and a governor or centrifugal force-responsive means for actuating said air control valve;

FIG. 2 is a diagrammatic view of the control system of FIG. 1, showing in cross-section the pneumatic-hydraulic transducer, the air pressure control valve therefor, and the centrifugal force-responsive means for actuating the air pressure control valve;

FIG. 3 is an enlarged fragmentary sectional view, taken along the line 3—3 in FIG. 2;

FIG. 4 is an end elevational view of the friction absorber as seen along the line 4—4 in FIG. 1, and showing a torque arm for actuating a pressure transmitting device connected to a gauge for indicating the torque being absorbed;

FIG. 5 is an enlarged vertical sectional view through the friction absorber, taken along the line 5-5 in FIG. 4, showing certain details of construction of the friction absorber, and the manner in which it is mounted in a coolant casing;

FIG. 6 is a graph comparing engine driving torque in foot pounds and friction absorber retarding force, with vehicle speed in miles per hour and showing in particular by the curves retarding force produced by controlling differently the pressure applied to the friction elements; and

FIG. 7 is a graph comparing road horsepower absorbed by the friction absorber with vehicle speed in miles per hour, and showing typical power curves for the friction absorber resulting from using the load control systems of the present invention.

DESCRIPTION OF A PREFERRED EMBODIMENT

Referring now to FIG. 1, a portion of a conventional chassis dynamometer for testing motor vehicles is shown for use in conducting under-load testing of the engine of the motor vehicle, the dynamometer including absorber assembly 2 to which driving torque is supplied by an input shaft 4 supported by spaced bearings 6. Driving torque is transmitted indirectly from the engine of the motor vehicle undergoing test to the input shaft 4 by a roll assembly 8, upon which the driving wheels (not shown) of the motor vehicle are supported.

The roll assembly 8 has a generally rectangular frame comprised of longitudinal side members 10, interconnected by transverse end members 12, only one of which is shown. The dynamometer includes two pairs of rolls, one of which pairs consists of the parallel rolls 18 and 20 mounted on shafts 22 and 24, respectively, which are supported by bearings 26 mounted on the transverse members 12. The shaft 22 extends beyond the end member 12, and is connected by a coupling 40 with the input shaft 4 of the power absorber assembly 2. Thus, when the engine of a motor vehicle positioned with one of its drive wheels disposed on the rolls 18 and 20 is operated to rotate said rolls, the wheel will drive the roll 18 to thereby transmit driving torque from the vehicle engine to the input shaft 4 of the power absorber assembly 2.

Referring now in particular to FIGS. 4 and 5, the power absorber assembly 2 includes a coolant-receiving casing 42. The front of the casing 42 is open and is surrounded by an external flange 52. A cover plate 54 and a gasket 55 are mounted on the flange 52 to close and seal the casing 42, and are secured to said flange by bolts 56. The casing 42 has an inlet pipe 57 connecting to an opening in the bottom wall 48 for admitting liquid coolant 58 into the casing. An outlet pipe 59 is connected to an opening near the top of the rear wall 44 of the casing for the discharge of said coolant.

The cover plate 54 has a centrally positioned boss 72 in which the input shaft 4 is rotatably mounted. The rear wall 44 of the casing 42 has a central boss 120 through which extends a bore 122 aligned with a bore 74 in a boss 72 on the cover plate 54.

Mounted within the casing 42 is a friction absorber 62A, which includes a rotor element or drum 64, and a

stator element 66, carrying movable brake shoes or friction pad elements 68. The drum 64 comprises a hub 96 having a socket 98, in which is keyed an enlarged end 82 of the shaft 4. A circular plate 108 and a gasket 109 are secured to a flange 106 on the drum 64 by cap screws 110.

The stator 66 comprises a cylindrical shaft portion 128, which projects through a boss 114 on a circular plate 130 of substantially smaller diameter than the inner diameter of the drum 64. The shaft 128 is rotatably mounted in the boss 114 and is also rotatably mounted in the boss 120 on the casing 42.

The friction pad means 68 comprises a pair of conventional arcuate brake shoes 158 and 160, each having brake lining material 162 secured thereto for frictionally engaging the inner cylindrical surface of the drum 64 when said shoes are moved outwardly, as will be readily understood. The plate 130 carries a pair of adjusting pins 154 upon which the brake shoes 158 and 160 are pivotally mounted.

Mounted on the plate 130 between the upper ends of the brake shoes 158 and 160 is a conventional fluid pressure operated brake actuator unit 176, which includes a cylinder 178 having a pair of piston-operated rods 180 and 182 extending from the opposite ends thereof, it being understood that the rods 180 and 182 are moved outwardly by fluid pressure within the cylinder 178 to expand the brake shoes 158 and 160. When fluid pressure is relieved in the cylinder 178, a return spring 174 functions to retract the brake shoes 158 and 160 out of engagement with the drum 64.

The shaft 128, FIG. 5, has an axial bore 184 which communicates at its inner end with an axial bore 185 and a radial bore 186 in the plate 130. The radial bore 186 communicates with the interior of the cylinder 178 for conducting fluid pressure to and from said cylinder. The outer end of the bore 184 is threaded to receive a fitting 188 to which a conduit 246 is connected. Thus, by supplying fluid under pressure through the conduit 246 the actuator unit 176 can be operated to move the friction pad means 68 into frictional engagement with the drum 64. The force with which such engagement is made will control the value of the resultant retarding force when the rotor or drum 64 is revolved. The magnitude of such force can be controlled by varying the value of the fluid pressure applied through conduit 246.

Mounted on the rear wall 44 of the coolant casing 42, FIGS. 4 and 5, and spaced from the shaft 128, is a transducer 192 for converting angular movement of the stator 66 into fluid pressure, said transducer including a housing 193 containing oil. An upwardly projecting push rod 194 extends from a piston 195 mounted in the housing 193. A conduit 196 full of oil is connected to the transducer 192, and leads to a fluid pressure operated gauge 198. The transducer 192 is constructed so that when the push rod 194 is depressed by mechanical force, hydraulic pressure will be produced by the piston 195 within the transducer and will be transmitted to the gauge 198 through the conduit 196. The hydraulic pressure will be relieved when the mechanical force applied to the push rod 194 is discontinued.

On the outer end of the shaft 128 is an arm 200 having a boss 202 fixed thereto by a key 204. The outer end 206 of the arm 200 rests on the push rod 194 of the transducer 192. When the brake actuator unit 176 is operated to move the friction pad means 68 into frictional engagement with the drum wall 104 and the rotor 64 is rotated counter-clockwise (as viewed in FIG. 4), the shaft 128 and the stator 66 will tend to rotate therewith. Angular movement of the stator 66 will engage the outer end 206 of the arm 200 with the push rod 194, producing a fluid pressure signal for transmission to the gauge 198. The value of the signal will be proportional to the effective retarding force of the friction absorber 62A and, hence, the gauge 198 can be calibrated accordingly in foot pounds.

When the rotor 64 is rotated while the friction pad means 68 is in engagement with the drum wall 104, heat will be generated between the relatively stationary friction brake shoes 162 and the moving surface of the drum wall 104. This is dissipated by passing coolant through the casing 42.

The load control apparatus for the friction absorber 62A includes the conduit 246 connected at one end to the fitting 188 for supplying fluid pressure to the brake actuator unit 176, the other end of the conduit 246 being connected to an air-pressure-hydraulic-pressure transducer 248, shown in corss-section in FIG. 2. The transducer 248 includes an upper housing section 250 and a lower housing section 252 having flanges 254 and 256, respectively, on their confronting ends and between which the outer margin of a flexible rolling diaphragm 258 is clamped. The housing section 250 has a hollow lower portion 260, and a reduced hollow upper portion 262, the latter terminating in a boss 264 to which the conduit 246 is connected.

Received within the housing sections 250 and 252 is a member 266 having a lower piston 268, and a relatively reduced upper plunger 270 slidably received within the portion 262 of the housing section 250. The plunger 270 carries a seal 272 in a groove 274 near its upper end. The lower face 276 of the piston 268 is engaged with the diaphragm 258, and the latter is secured thereto by a bolt 278 and a washer 280. The conduit 246 and the chamber in the housing portion 262 above the plunger 270 are filled with a suitable hydraulic fluid 282. Thus, when the member 266 is moved upwardly, the fluid 282 will be pressurized by the plunger 270 for operating the brake actuator unit 176.

The chamber in the lower housing section 252 has a port 284 communicating therewith, to which is connected one end of a conduit 286 leading from an air pressure control valve 288 connected to an air pressure source 290. When air pressure is supplied to the housing section 252 beneath the rolling diaphragm 258, the piston 266 will be moved upwardly to exert force on the fluid 282. The area 276 of the piston 266 against which air pressure acts through diaphragm 258 is several times greater than the area of the upper end face 292 of the plunger 270, so that the pressure on the surface 276 will be correspondingly multiplied in the fluid 282.

The control valve 288 is operable mechanically and automatically to control the value of air pressure transmitted from the source 290 to the lower section 252 of the transducer 248. The valve 288 includes right and left housing sections 294 and 296 (as viewed in FIG. 2) between the confronting ends of which a rolling diaphragm 298 is clamped. The housing section 296 has a valve chamber 300 extending from the end face

302 thereof, said valve chamber including a frusto-conical seat 304 at its bottom, and terminating in a passage 306 leading to a larger chamber 308. The chamber 308 has a frusto-conical side wall portion 310, which terminates at a shoulder 312, and faces the diaphragm 298. A circular valve seat 314 is secured to the shoulder 312. An inlet port 318 communicates with the central portion of the valve chamber 300, and one end of a conduit 320 extending from the pressure source 290 is connected thereto. An outlet chamber 322 leads from chamber 308 and the region of the passage 306 to an outlet port 324, to which one end of the conduit 286 is connected.

Received within the valve chamber 300 is a valve 326 having an enlarged head 328 with a hemi-spherical surface engageable with the seat 304 to close the passage 306, and a stem 330 which extends through the passage 306 and through a central opening in the seat 314. The outer end of the valve chamber 300 is closed by a plug 334 held in position by a plate 336 secured to the housing section 296 by screws 338. A spring 340 is compressed between the head 328 of the valve 326 and the plug 334, and functions to urge the spherical surface on said head into seating engagement with the seat 304.

The diaphragm 298 has a central opening therein, through which projects the threaded end of a flanged member 342 having an axial passage 344, one end of which is frusto-conical to provide a seat 345 for receiving the tip of the stem 330. The size of the passage 344 is chosen so that when the tip of the stem 330 contacts the seat 345, said passage will be closed. A diaphragm support member 348 having a central boss is threaded on the member 342 and secures it to the diaphragm 298.

A cup-shaped member 352 is fitted over the boss on the member 348 and has a wall 354 spaced from the end face of the members 342 and 348. A plurality of circumferentially spaced passages 356 extend through the wall 354. The housing section 294 has a vent port 360 in the wall thereof, whereby air under pressure flowing through the passage 344 will travel through the passages 356, and will exhaust through the vent port 360.

The end wall 366 of the housing section 294 has a boss 368 into which a flanged guide 372 is threaded. The guide 372 has an axial bore 374 through which a push rod 364 extends. The member 352 has a boss on the end thereof within which a socket is provided for the adjacent end of the push rod 364. The push rod 364 is actuated by a mechanical linkage under the control of a speed responsive device driven by a belt 376 from the pulley 220. The speed responsive device includes a flyweight centrifugal governor unit 378, comprising a cylindrical housing 380 having a reduced extension 382 at one end within which a pair of spaced ball bearings 384 is mounted. A shaft 386 rotates in the bearings 384, and has a pulley 388 secured to its outer end to be driven by the belt 376. Thus, the shaft 386 will be rotated at a speed directly proportional to the speed of the power input shaft 4.

The inner end of the shaft 386 has a cross member 390 from the ends of which extend axial supports 392. One end of a fly-weight 394 is pivoted to each support 392 by a pin 396, so that when the shaft 386 is rotated, the free ends of said fly-weights will be moved outwardly by centrifugal force, as will be readily understood. Each of the weights 394 carries a lug 398 which will move axially away from the shaft 386 as the weights 394 swing outwardly upon rotation of said shaft.

The open end of the housing 380 is closed by a plate 400 having a central boss 402 containing a bore 404. A cylindrical bushing 406 is fixed in the bore 404, and slidably receives the stem 408 of an actuator element 410 having a head 412 adjacent to which is attached a ball thrust bearing 414. The lugs 398 engage one race of the bearing 414 in order to allow relative rotation between said lugs and the element 410.

When the shaft 386 is rotated and the weights 394 swing outwardly, the lugs 398 function to push the element 410 axially forwardly out of the housing 308. Thus, the centrifugal unit 378 functions to convert rotary movement of the shaft 386 into linear movement of the stem 408. The fly weights 394 and their lugs 398 are designed so that the element 410 will be moved axially in direct relationship to centrifugal force acting on the weights 394. Further, it is known that centrifugal force changes as the square of the rotational speed. Thus, the element 410 will be shifted axially according to the square of the changes in speed of the shaft 386.

Movement of the element 410 is transmitted to the push rod 364 through a parallel lever arrangement 416, including a first lever 418, pivotally mounted at its upper end on a fixed pin 420 positioned in the same plane as the element 410. A second lever 422 extending parallel to the first lever 418, and pivotally mounted at its lower end on a fixed pin 424 is also disposed in the same plane as the push rod 364. The lower end of the lever 418 is engaged with the outer end of the push rod 364, and the upper end of the lever 422 is engaged by the outer end of the element 410.

The levers 418 and 422 are spaced apart, and received there between is an adjustable fulcrum wheel 426 having guide flanges 428 for retaining the levers 418 and 422 engaged with its outer surface. The fulcrum wheel 426 is carried by a yoke 430, the lower end of which is connected by a conventional universal joint 432 to a swivel head 434.

The swivel head 434 has a bore 436 in the underface thereof, within which a ball bearing 438 is secured by a snap ring 440. The reduced upper end 442 of a threaded rod 444 passes through the bearing 438 and is rotatably secured thereto by a snap ring 446. The rod 444 also extends through a threaded opening in a fixed plate 448, and has a hand wheel 450 mounted on its lower end. Thus, by turning the hand wheel 450 the fulcrum wheel 426 can be adjusted along the levers 418 and 422 from a position opposite the upper pivot pin 420 to a position opposite the lower pivot pin 424.

Depending upon where the fulcrum wheel 426 is positioned, the lever arrangement 416 can multiply or divide the overall axial movement of the element 410 and effect a proportionate movement of the push rod 364, and which movement in any event, is correlated to the speed of the shaft 386.

For example, if the fulcrum wheel 426 is positioned directly opposite the upper pivot pin 420, no force would be transmitted to the push rod 364 by the element 410 because then no outward movement of said

element could occur. The air control valve **288** would therefore remain closed during rotation of shaft **386**, and no retarding action would be produced by the power absorption unit **2**.

If the adjustable fulcrum wheel **426** is positioned midway between the pivot pins **420** and **424**, the movement transmitted to the push rod **364** will be directly proportional to the force exerted on the lever **422** by the element **410**. This force is then converted to a proportional air signal by the air control valve **288**, and is transmitted to the air pressure-oil pressure signal by the ratio of the area **276** of the piston **268** exposed to air pressure, to the area **292** of the plunger **270** exposed to the hydraulic fluid **282**, and produces fluid pressure for transmission through the conduit **246** to the brake actuator unit **176** of the friction absorber **62A**. As the speed of the prime mover, and hence of the rotor **64** of the friction absorber **62A** is changed, the signal pressure to the brake actuator unit **176** will be changed with the square of the change in speed. The result is, that the retarding force produced by the friction absorber **62A** will have the characteristics of the curve D in FIG. **6**, and the power curve for the friction absorber will be as shown at H in FIG. **7**.

If the adjustable fulcrum wheel **426** is positioned closer to the lower pivot pin **424** than to the upper pivot pin **420**, the force or movement transmitted to the push rod **364** is multiplied, resulting in a higher air pressure being transmitted to the transducer **248**, and in a greater retarding force exerted by the power absorber unit **62A**. Similarly, if the movable fulcrum wheel **426** is positioned nearer the upper pivot pin **420** than the lower pivot pin **424**, the signal pressure to the brake actuator unit **176** will be less than when said fulcrum wheel is positioned midway between the pivot pins **420** and **424**. The force exerted on the movable friction pad elements **68** by the brake actuator unit **176** can thus be easily varied anywhere between zero and maximum for any given speed of the input shaft **4**, merely by adjusting the position of the fulcrum wheel **426**.

In operation, when no inward pressure is exerted on the push rod **364**, the head **328** of the valve **326** will be held in engagement with the seat **304** by the coil spring **340**, thus closing passage **306**. No air pressure will then flow from the conduit **320** into the conduit **286**.

When the push rod **364** is moved inwardly, the head **328** will be disengaged from the seat **304** and an annular passage from the valve chamber **300** to the passage **306** will be established. The size of this annular passage, and hence the rate of air flow through the passage **306** will vary with the extent to which the valve **326** is opened, said valve being designed so that the change in area of said annular flow space will be directly proportional to the inward movement of the push rod **364**.

When the pressure on the push rod **364** is relieved, and said push rod is allowed to move outwardly, the spring **340** will return the valve **326** to seating engagement with the seat **304** for closing the passage **306**. Thereafter, air pressure returned from the transducer **248** to the housing section **296** through the conduit **286** will act on the diaphragm **298** to move the same outwardly, thereby unseating the end of the valve stem **330** from the seat **345** at one end of passage **344**. Air pressure will then exhaust to atmosphere through the passage **344**, the passages **356**, and the vent port **360**, whereby air pressure on the bottom face of the piston **268** of the transducer **248** will be relieved. It is thus seen that by manipulating the push rod **364**, the pressure exerted by the brake actuator unit **176** to engage the friction pad means **68** can be varied at will, and that for each longitudinal position of the push rod **364** there will be a corresponding air pressure established in the transducer **248**, resulting in a corresponding fluid pressure being transmitted to the brake actuator unit **176**.

Referring now to the graph of FIG. **6**, the driving torque-speed characteristics for a typical motor vehicle engine is indicated by the curve A. Here, the values of torque in foot pounds are plotted as ordinates, and the corresponding vehicle speeds in miles per hour are plotted as abscissas. It is seen that the engine driving torque rises rapidly with increasing engine speed, from 0 to about 25 miles per hour, and that thereafter driving torque increases at a slower rate with engine speed, until at about 47 miles per hour the driving torque becomes stabilized. Over the range from about 47 miles per hour to about 65 miles per hour, no appreciable increase in driving torque occurs. Above about 65 miles per hour, the value of engine driving torque decreases with increasing engine speed.

One manner of operating the friction absorber **62A** would be to supply a constant fluid pressure to the brake actuator **176**, the result of which is illustrated by the curve B in FIG. **6**, wherein it is seen that the value of the retarding force would then be constant over the complete range of engine speed, from 0 miles per hour upwardly.

While under-load testing a motor vehicle engine, or other prime mover, it is desirable to operate the engine at several different stable operating speeds. When using friction dynamometer equipment, such a stable speed is obtained by matching the value of the generated retarding force to the value of the driving torque, until operation of the prime mover at the desired preselected speeds results. Turning to the curves A and B in FIG. **6**, it is seen that between about 47 and about 65 miles per hour the engine driving torque curve A is parallel with the constant value retarding force curve B. Because of this parallel relationship, it is practically impossible over this common driving speed range to match retarding force to the driving torque and effect stability. The result is a "hunting" action, or a "running wild" of the engine, and hence true performance testing of the engine is not possible.

Another problem, with a constant retarding force, results from the fact that in the lower speed ranges, driving torque decreases rapidly in value with decreased speed, as is shown by the curve A, FIG. **6**. Thus, assuming that under-load testing is in progress at a substantially stable engine speed of 30 miles per hour, a problem arises if the engine should misfire or momentarily lose power. While there would then be an immediate decrease in driving torque, the retarding force would remain constant, and as the driving torque began to decrease, the constant retarding force would act to further slow the engine, and rapid decrease in driving torque would occur until the engine completely stalled. This condition can be alleviated by having the retarding force increase and decrease with changes in speed, and hence with driving torque.

It has been found that for the most efficient engine operation, the value of the retarding force should preferably be substantially zero at zero engine speed, and should rise and fall faster than the changes in the driving torque output of the prime mover being tested. When the value of the retarding force is thus varied, the power absorbed, versus engine speed, will increase and decrease more rapidly than engine power output.

Referring again to FIG. **6**, curve C represents a situation where retarding force is varied directly with changes in engine speed, which can be done by varying the value of the fluid pressure supplied to the brake actuator unit **176** in direct proportion to changes in the speed of the engine being tested.

Thus, at zero speed the retarding force is also zero. As the engine speed increases, there is a corresponding increase in driving torque, and similarly, when engine speed decreases the retarding force changes accordingly. It is seen that the retarding force curve C cuts sharply across the typical driving torque curve A, at about **58** miles per hour, and that there are no regions where the retarding force curve C is parallel with the engine driving torque curve A. Thus, retarding force can easily be matched with driving torque to provide a stable operating speed, and there is no problem of engine stall occurring when there is a temporary decline in driving torque, because the retarding force follows such decline.

In the case of automotive engines, it has been found that the best relationship for retarding force is to have the value thereof increase and decrease as approximately the square of the change in speed, and to be 0 at zero speed. The reason this is a nearly ideal condition is that it very closely simulates the load actually imposed on a conventional automobile engine while the vehicle is being driven on a level road. Such a retarding force versus speed curve is shown at D in FIG. **6**. Such retarding force can be created by varying the value of the pressure on the friction pad elements **162** in accordance with the square of the engine speed.

The load control apparatus or system of the present invention will vary, with changes in speed, the pressure with which the friction pad elements **162** are urged into frictional engagement with the drum wall **104** of the rotor **64**. The system, therefore, is effective to vary the value of the retarding force produced in proportion to the driving torque. The load control apparatus is designed so that the retarding force versus speed curve of the friction absorber **62A** can be shifted to the right or left around zero in FIG. **6** to obtain nearly any desired value of retarding force at any given speed, whereby nearly any stable operating speed can be established for under-load testing of a prime mover.

FIG. **7** is a graph wherein road horsepower is plotted against vehicle speed in miles per hour, the curve E showing a typical power curve for an automobile engine. A typical power curve for a friction absorber wherein the retarding force is constant is shown at F, and it is seen that the slope of the curve F is substantially less than that of the curve E, whereby the power absorbed by the friction absorber **62A** rises and falls at a slower rate with speed than does engine power. On the other hand, the curve H plots the power absorbed by the friction absorber **62A** against engine speed in terms of vehicle speed in miles per hour, and it is seen that in this instance the absorbed power curve has a slope substantially greater than the engine power curve E, whereby the absorbed power rises and falls at a rate faster than the increases and decreases in engine power. The arrangement of FIGS. **1** to **5** thus make it possible to easily attain any desired stable operating speed for a prime mover during under-load testing, and because retarding force and absorbed power rise and fall at faster rates than driving torque and engine power, rapid response of the friction absorber **62A** to changes in vehicle speed is assured and the problem of stalling in instances where the prime mover momentarily loses power is eliminated.

Under certain conditions it is possible to substitute a pneumatically operated brake actuator unit in place of the hydraulic brake actuator unit **176**, FIG. **1**, for moving the friction members **162** into engagement with the rotor drum **64**. If a pneumatic brake actuator unit is substituted for the hydraulic actuator unit **176**, then the air-pressure fluid-pressure transducer **248** can be eliminated, and the friction elements **162** would then be operated directly by the air pressure output from the valve **288**.

Obviously, many additional modifications and variations of the present invention are possible in the light of the above teachings.

I claim:

1. Load control means for controlling the retarding force characteristics of a rotary power absorber for absorbing driving torque from a shaft while said shaft is being driven from a prime mover, said power absorber including means for producing retarding force, and fluid pressure responsive means for causing said retarding means to produce a retarding force to oppose said driving torque, comprising: speed responsive pressure regulating means including a governor to be driven from the prime mover for providing a fluid pressure corresponding in value to, and varying as, the speed of rotation of the prime mover and including an air pressure control valve connected with a source of supply of air under pressure, said air valve being actuated by said speed responsive means, and means including an air-pressure-to-hydraulic-pressure transducer interposed between the air pressure control valve and the means for producing the retarding force for transmitting the controlled fluid pressure from said air pressure control valve to said fluid pressure responsive means to apply a retarding force of a corresponding value to the power absorber.

2. Load control means for controlling the retarding force characteristics of a rotary power absorber for absorbing driving torque from a shaft while said shaft is being driven from a prime mover, said power absorber including means for producing retarding force, and fluid pressure responsive means for causing said retarding means to produce a retarding force to oppose said driving torque, comprising: speed responsive pressure regulating means including a governor to be driven from the prime mover for providing a fluid pressure corresponding in value to, and varying as, the speed of rotation of the prime mover and including an air pressure control valve connected with a source of supply of air under pressure, an adjustable linkage disposed between said governor and said valve, actuated by said speed responsive means, and means for transmitting the controlled fluid pressure from said valve to said

fluid pressure responsive means to apply a retarding force of a corresponding value to the power absorber.

3. Load control means as defined in claim 2, wherein the adjustable linkage comprises a pair of generally parallel bars with a fulcrum member therebetween, one bar being pivotally mounted at one end and the other pivotally mounted at the end thereof remote from said one end, and wherein the governor includes a movable element acting on the free end of one of said bars, and the fluid pressure control valve includes a member acted upon by the free end of the other of said bars.

4. Load control means as defined in claim 3, wherein the fulcrum member between the bars is a roller, and means is connected with said roller for adjusting its position along the length of said bars.

5. Load control means for controlling the retarding force characteristics of a rotary power absorber for absorbing driving torque from a shaft while said shaft is being driven from a prime mover, said power absorber including means for producing retarding force, and hydraulic fluid pressure responsive means for causing said retarding means to produce a retarding force to oppose said driving torque, comprising: speed responsive pressure regulating means to be driven from the prime mover for providing a fluid pressure corresponding in value to, and varying as, the speed of rotation of the prime mover and including fluid pressure control means actuated by said speed responsive means, said fluid pressure control means including a flow control valve having an inlet and an outlet, a source of air supply under pressure connected to the flow control valve inlet, and means for transmitting the controlled air pressure from the outlet of said flow control valve to said hydraulic fluid pressure responsive means to apply a retarding force of a corresponding value to the power absorber and including an air-pressure-to-hydraulic-pressure transducer for transforming the controlled air pressure to hydraulic pressure.

6. Load control means as defined in claim 5, wherein the speed responsive pressure regulating means includes a governor device operable in response to variations in the speed of the prime mover; a linkage actuated by force supplied by said governor; and an adjustable fulcrum element for increasing or decreasing the force developed by said governor, the flow control valve being arranged to be actuated by said linkage in proportion to the force exerted thereon by said governor.

7. Load control means for controlling the retarding force characteristics of a friction type rotary power absorber for absorbing driving torque from a shaft while said shaft is being driven from a prime mover, said power absorber including friction brake means for producing retarding force, and fluid pressure responsive means for causing said friction brake means to produce a retarding force to oppose said driving torque, comprising: speed responsive pressure regulating means to be driven from the prime mover for providing a fluid pressure corresponding in value to, and varying as the speed of rotation of, the prime mover and including a source of air pressure; a conduit connected with said source of air pressure; an air pressure control valve connected with said conduit for controlling the air pressure from said air source, said air pressure control valve being operated in response to the speed of rotation of the prime mover to control the fluid pressure to said fluid pressure responsive means to actuate said friction brake means, the value of said controlled air pressure being a pre-selected function of the speed of rotation of the prime mover; and means for transmitting the controlled fluid pressure from said air pressure control valve to said fluid pressure responsive means to apply a retarding force of a corresponding value to the power absorber.

8. A friction type power absorber for use in analyzing the performance of a prime mover, comprising: a driven brake shaft for receiving the driving torque from a prime mover; power absorption means including rotor means connected to receive driving torque from said shaft, stator means operatively disposed relative to said rotor means; friction brake means carried by one of either said rotor or stator means and movable into and out of frictional engagement with the other; and fluid pressure operated actuator means operable to apply force for moving said friction brake means into said frictional engagement to thereby apply retarding force to said rotor in opposition to driving torque applied thereto by said shaft, the value of said retarding force varying with the force exerted by said actuator means; and control means including means to generate a pressure signal to actuate an element of a fluid system for controlling said actuator means by fluid pressure regulated in accordance with the speed of rotation of said shaft so that the force exerted on said friction brake means by said actuator means varies as a function of said rotational speed and so that said retarding force is substantially zero at zero speed of said shaft and increases and decreases in value at a rate faster than said driving torque.

9. A friction type power absorber as defined in claim 8, wherein the value of the retarding force is varied in accordance with a mathematical function of the speed of rotation of the prime mover.

10. A friction type power absorber as defined in claim 8, wherein the value of the retarding force is varied in accordance with substantially the square of the speed of rotation of the prime mover.

11. A friction type power absorber as defined in claim 8, wherein the control means includes a speed responsive governor.

12. A friction type power absorber as defined in claim 11, wherein the governor is driven at a speed proportional to the speed of the brake shaft driven from the prime mover.

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